Static and dynamic characteristics of direct operated pressure relief valves

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Static and dynamic characteristics of direct operated pressure relief valves: *This paper is dealing with static and dynamic performances of direct operated pressure relief valves. The subject of research was Rexroth pressure relief valve type DBD6 A6G 200. A mathematical model of the valve was obtained and an algorithm was developed in MATLAB. Experimental test stand was built for experimental verification of the theoretical mathematical model. Theoretically end experimentally obtained static and dynamic characteristics of the valve are presented in few diagrams.*

Keywords*: direct operated pressure relief valve, flow, pressure, static characteristics, dynamic characteristics, transient response, mathematical model, experiment.*

INTRODUCTION

Many authors [1], [2], [3] have investigated the transient response characteristics of hydraulic systems with direct operated pressure relief valves. But there are not presented simplified expressions for determination of the pressure peak and oscillation frequency of the transient response.

Activating the directional control valves in the hydraulic systems with direct acting pressure relief valves it occurs a transition process in which it is possible the pressure to reach values several times higher than the steady state value. This occurs overload that can cause undesirable consequences.

Experimentally and theoretically determined transient response of hydraulic systems is presented in this paper and mathematical expressions for quality determination of the transient response are shown, as well. This will allow the designers to assess in advance the dynamics of the created system.

FUNCTIONAL DIAGRAM OF THE VALVE

The investigated pressure relief valve is presented on fig.1. On *fig.2* a functional diagram of the test stand with direct acting pressure relief valve with dumping piston, volume of oil at its inlet V_0 and output pipeline with linear R_p and inertial L_p resistance is shown. To isolate the oil compressibility between the pump and the valve and for reducing pressure pulsation of the pump, it is included a throttle with high inertial resistance.

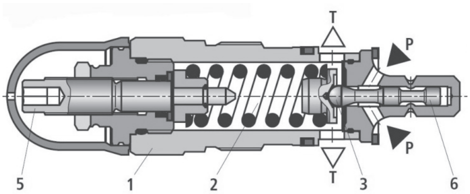

Fig.1. The investigated pressure relief valve

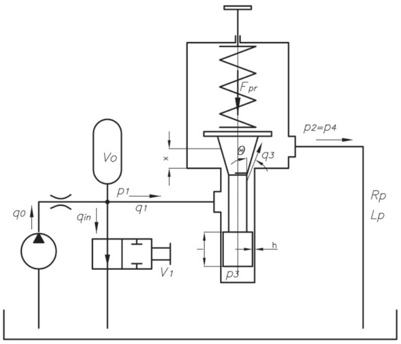

Fig.2. Functional diagram of the test stand

A rapid closure of the opening x_v of the directional control valve $V1$ with plunger diameter d_v, creates a transient response. Pump flow q_0 enters in the inlet volume V_0 and the inlet pressure p_1 is increasing. The valve opens when the set pressure p_0 is reached and through the outlet pipeline the oil q_1 flows back in the tank.

MATHEMATICAL MODEL OF THE TRANSIENT RESPONSE

Mathematical model of the system is described by the following equations:

- *Equation of continuity in front of the valve*

$$q_0 = q_{in} + q_v + q_1 \tag{1}$$

where: $q_{in} = \left(1 - \frac{t}{t_1}[\square)\mu\right]_v \pi d_v x_v \sqrt{\frac{2}{\rho} p_1}$ -flow through the directional control valve *V1*, which closes for time t_1; t – time; μ_v – flow coefficient through the directional control valve; $q_v = \frac{V_0}{K}\frac{dp_1}{dt}$ – flow which enters in the volume V_0; $q_1 = \mu\pi d x \sin\theta \sqrt{\frac{2}{\rho} p_{1,2}}$ –flow through the valve with diameter d, angle of flowing θ, opening x and pressure drop $p_{1,2}$.

- *Equation of continuity in the valve in front of the control orifice and after it*

$$q_1 = q_2 = q_3 + A_k \frac{dx}{dt} \tag{2}$$

where: A_k-area of the closing element of the valve; q_3 –flow through the control orifice in the valve.

- *Equation of motion of the closing element of the valve*

$$m\frac{d^2x}{dt^2} + c(h_0 + x) + r_h x p_{1,2} = A_k(p_3 - p_4) - F_T \tag{3}$$

where: $m = m_k + \frac{1}{3}m_f$ – equivalent mass of the closing element m_k and the spring m_f ; c – stiffness of the spring; h_0 – deformation of the spring when $x=0$; $r_h = \mu\pi d.\sin 2\theta$ – coefficient of the hydrodynamic force; F_T –friction force between the closing element and the body of the valve.

The pressure in the lower region of the closing element of the valve p_3 depends of the losses in the orifice between the piston of the closing element and the body of the valve:

$$p_3 = p_1 - R_{a,l}A_k\frac{dx}{dt} - R_{a,m}\left(A_k\frac{dx}{dt}\right)^2 - L_a A_k \frac{d^2x}{dt^2} \tag{4}$$

Where: $R_{a,l}$, $R_{a,m}$ and $L_a = \rho\frac{l}{\pi d h}$ are linear, local and inertial resistances in the orifice with length l.

The pressure in the upper region of the closing element is obtain analogically when for this type of the valve is $p_4=p_2$.

- Equation of flowing in the outlet pipeline

$$p_2 = R_{p,l}q_2 + R_{p,m}q_2^2 + L_p\frac{dq_2}{dt} \tag{5}$$

Where: $R_{p,l}, R_{p,m}$ and $L_{p,t}$ respectively linear, local and inertial resistance of the outlet pipeline with length l_p and diameter d_p.

If all the differentials (the variables dependent of the time) in the equation (1-5) are set to zero, the mathematical model of the static characteristic of the valve is obtained.

EXPERIMENTAL AND THEORETICAL CHARACTERISTICS

The measurement instruments were previously calibrated. A pressure transducer type HM17 manufactured by BoschRexroth was used for pressure measurement. For displacement of the valve a position sensor type 1 837 001 304 manufactured by BoschRexroth was used. The data are stored in the computer through 14 bit data acquisition card NI USB-6009 manufactured by National Instruments.

The theoretical and experimental static characteristic for different pressure setting is presented at fig.3. At the same figure, the displacement of the valve is shown, as well.

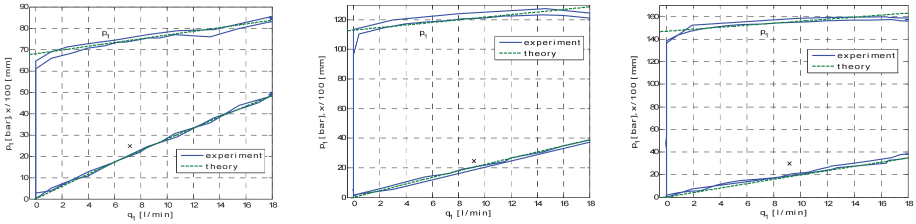

Fig.3 Experimental and theoretical static characteristic

For transient response of the specified valve a steady state flow of 25 l/min was set. Different experiments for different volume in front of the valve were made. The results of the experimental investigation and the solution of the theoretical model are presented at *fig.4*.

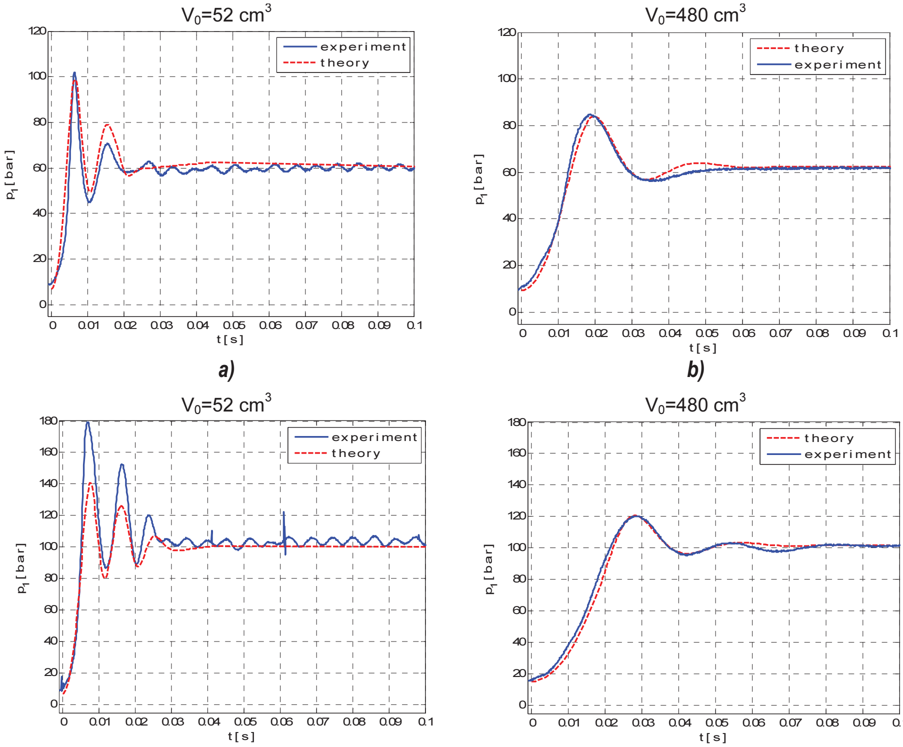

Fig.4 Experimental and theoretical dynamic characteristics for different values of the pressure and volume at the inlet port

As can be seen at *fig.4*, a very fast response for lower inlet volume of oil is obtained. For inlet volume of 52 cm^3 the settling time is around 20 ms for pressure of 60 [bar] and around 30 ms for pressure of 100 [bar]. For higher inlet volume of oil the transient response is slower, the settling time is around 50 ms for pressure of 60 [bar] and around 75 ms for pressure of 100 [bar]. The pressure overshooting and frequency of oscillation is different for different inlet volume of oil, as well. Determination of the pressure overshooting and the dumped natural frequency of the valve are presented in the next paragraph with the linearization of the non-linear model (1-5).

LINEAR APROXIMATION OF THE TRANSIEN RESPONSE

Presented transient responses at *fig.4* have the form of the transient response of a linear second-order system. The parameters of this system can be approximately determined if it is analyzed the linear model of the system around the steady state values of the parameters p_0 and q_0. On *fig. 5* it is shown a structural diagram of the investigated system. The transient response of closing of the directional control valve $V1$ is neglected. $P_{1,2}$, Q_0, Q_1, P_2 и Q_v are Laplace dimensionless parameters to the relative coordinates – pressure drop in the valve $\frac{\Delta p_{1,2}}{(p_{1,2})}$, inlet flow $\frac{\Delta q_0}{q_0}$, flow through the valve $\frac{\Delta q_1}{q_0}$, outlet pressure $\frac{\Delta p_2}{(p_{1,2})_0}$ and flow into the compressible volume of oil at inlet port $\frac{\Delta q_v}{q_0}$.

The transfer function W_k of the valve, presented in [4], is:

$$W_k = \frac{P_{1,2}}{Q_1} = k_{pq}\frac{T_k^2 s^2 + 2\xi_k T_k s + 1}{T_0^2 s^2 + 2\xi_0 T_0 s + 1} \tag{6}$$

where $k_{pq} = k_{st}\frac{q_0}{(p_{1,2})_0}$ is the slope of the dimensionless static characteristic of the valve; $T_k = \sqrt{\frac{m + L_a A_k^2}{c + r_h (p_{1,2})_0}}$ and $\xi_k = \frac{R_a A_k^2}{2T_k\left[c + r_h (p_{1,2})_0\right]}$ - time constant and relative damping coefficient of the closing element of the valve; $T_0 = \frac{T_k}{\sqrt{1 + 2k_{x,p}}}$ and $\xi_0 = \frac{\xi_k T_k + k_{x,p} T_a}{T_k\sqrt{1 + 2k_{x,p}}}$ - time constant and relative damping coefficient of the control system of the valve; $k_{x,p} = \frac{A_k (p_{1,2})_0}{\left[c + r_h (p_{1,2})_0\right] x_0}$ and $T_A = \frac{A_k x_0}{q_0}$ are respectively opening coefficient of the valve and time constant; s- Laplace operator.

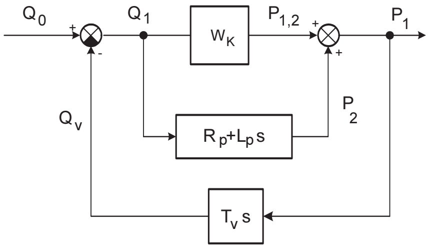

Fig.5 Block diagram of the linear model

The outlet flow from the valve is flowing through the pipeline with linear resistance in dimensionless coordinates $R_p = R_{p,l}\frac{q_0}{(p_{1,2})_0}$ and inertial resistance $L_p = L_{p,r}\frac{q_0}{(p_{1,2})_0}$:

$$P_2 = R_p Q_1 + L_p s Q_1 \tag{7}$$

The inlet pressure P_1 changes the flow Q_v, which enters in the volume V_0 according to the expression:

$$Q_v = T_v s P_1 \tag{8}$$

where $T_v = \frac{V_0}{K}\frac{(p_{1s})_0}{q_0}$ is a time constant of the compressible volume of oil.

Solving the equations (6), (7) and (8), the transfer function of the whole system $W_o = \frac{P_1}{Q_o}$ is obtain:

$$W_o = \frac{a_1 + a_2 s + a_3 s^2 + a_4 s^3}{b_1 + b_2 s + b_3 s^2 + b_4 s^3 + b_5 s^4} \tag{9}$$

where the coefficients a_1, a_2, a_3, a_4 and b_1, b_2, b_3, b_4 and b_5 are directly obtain from the equations (6), (7) and (8). An analysis of the characteristic polynomial shows that the values of the roots are strongly different, so it is possible to approximate the transient response of the system with second order system. The values of the coefficients b_1, b_2, b_3 for the simplified system are:

$$\begin{aligned} b_1 &= 1 \\ b_2 &= 2\xi_0 T_0 + k_{pQ} T_v + R_p T_v \\ b_3 &= T_0^2 + k_{pQ} 2\xi_k T_k T_v + R_p 2\xi_0 T_0 T_v + L_p T_v \end{aligned} \tag{10}$$

From these values it is possible to calculate the frequency of oscillation of the system $\omega_s = \frac{\sqrt{(1 \quad \xi_s^2)}}{\sqrt{b_3}}$ and the overload $p_{1,max} = p_0 \left(1 + e^{-\frac{\xi_s \pi}{\sqrt{1-\xi_s^2}}}\right)$. Here $\xi_s = \frac{b_2}{2\sqrt{b_3}}$ is a relative damping coefficient of the approximated second order system.

In the following *Table 1* the values of the overload and the frequency of the oscillation are shown from the experiments, the theoretical nonlinear model and the approximated linear model (10) if the inlet flow is *25 l/min*. The time of closing of the directional control valve *V1* is around *0.018 s*.

Table 1.

	p_0=60 bar, V_0=52 cm^3	p_0=60 bar, V_0=480 cm^3	p_0=100 bar, V_0=52 cm^3	p_0=100 bar, V_0= 480 cm^3
Experiment	$p_{1,max}$=100 bar ω_s=742 rad/s	$p_{1,max}$=85 bar ω_s=206 rad/s	$p_{1,max}$=180 bar ω_s=709 rad/s	$p_{1,max}$=120 bar ω_s=225 rad/s
Non-linear model	$p_{1,max}$=98 bar ω_s=778 rad/s	$p_{1,max}$=84 bar ω_s=227 rad/s	$p_{1,max}$=140 bar ω_s=650 rad/s	$p_{1,max}$=120 bar ω_s=217 rad/s
Approximated model	$p_{1,max}$=72 bar ω_s=689 rad/s	$p_{1,max}$=82 bar ω_s=257 rad/s	$p_{1,max}$=128 bar ω_s=679 rad/s	$p_{1,max}$=135 bar ω_s=272 rad/s

As can be seen from the above table the approximated model deviates maximum of around 20% from the experimental data. This is due to relatively slow closing of the directional control valve *V1*, which reflects the results. The data of the experimental and non-linear model match well.

CONCLUSION

The approximated model allows simple calculation of the parameters of the overload. As can be seen, the direct acting pressure relief valve with dumping piston shows a relatively large amount of dynamic overload of the hydraulic systems. In cases where this overload cannot be accepted it is advisable to use pilot operated pressure relief valves.

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The paper is reviewed.